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Mechanical Pumped Cooling Loop for Spacecraft Thermal Control

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ABSTRACT

The Mars Pathfinder (MPF) Spacecraft, scheduled for a December '96 launch to Mars, uses a mechanically pumped loop to transfer dissipated heat from the insulated lander electronics to an external radiator. This paper discusses the tradeoffs performed before choosing a mechanical pumped loop as the therms 1 control system for MPF. It describes the analysis, design, and predicted performance of this system. Tradeoffs performed in the selection of the working fluid, tubing diameters and materials, layout of the tubing and the working fluid venting approach arc described. The various development tests performed arc discussed along with the current status of this cooling system. Finally, some thoughts on the development of mechanically pumped loops for future spacecraft arc presented.

1. INTRODUCTION

The Mars Pathfinder (MPF) Spacecraft is scheduled for a December '96 launch to Mars with a landing in July '97. The spacecraft (Figure 1) is composed of two distinct parts: the cruise stage and the lander. The cruise stage is separated from the lander by several explosive bolts, just prior to the lander portion entering the Martian atmosphere. Safe entry of the lander into the Martian atmosphere is achieved by an ablative heat shield surrounding it.. Upon reduction of the spacecraft speed to a predetermined level, the heat shield is detached and discarded followed by deployment of a parachute to slow it clown further. At a predetermined low altitude, this is followed by firing of rctro-rockets to bring the lander to a near standstill. Finally, airbags surrounding the lander system are deployed to cushion the impact on the Martian surface.

Following landing the airbags arc vented and retracted and the tetrahedral shaped lander deploys its self righting petals which exposes the three solar arrays and the insulated electronics enclosure. Finally, a rover located on onc of the solar panels is released and allowed to discover the landscape of the Martian surface.

The same communication and data analysis electronics is used both during cruise and landed operations. It is located in the lander basepetal and is completely enclosed in a very high performance insulation to conserve heat. (power) during the cold Martian nights (as cold as -80°C). Because the same electronics is used both during cruise as well as landed, and since it is highly insulated (insulation and stowed airbags), it is very difficult, to remove its dissipated heat passively during the cruise phase (temperatures outside the insulated enclosure areas high as 15°C near earth). Further, the 90 Watts of power is continuously dissipated by this lander electronics during cruise. This necessitated the need for a heat reject.ion system (HRS) for MPF. In addition to rejecting heat during the cruise phase the HRS was also required to minimize any heat, leaks from the insulated electronics once MPF has landed on Mars,

2. SELECTION OF MECHANICAL PUMPED COOLING LOOP FOR HRS

'l'he 1 IRS serves as the thermal link from the equipment shelf to the heat sink - it picks up the heat from the electronics and transfers it to the radiator which in turn rejects it to space via thermal radiation. 'l'he heat shield and a cruise stage radiator were investigated as possible heat sinks for the 1 11{S. The concepts traded-off for the HRS were pumped fluid loops (single phase, liquid), heat pipes (variable & constant conductance, VCHP and CCHP) and detachable thermal/mechanical links. Mass, cost, schedule, power & technology readiness were traded. A mechanical pumped fluid cooling loop using freon- 11 was chosen for the HRS due to the following attractive features (which other concepts lacked partially or fully):

- Ease of integration with s/c
- Easily severed link before Martian entry (pyro cutting of tubes)
- No need for power during Martian nights (unlike VCH l')
- Flexibility of ground operations & tests (no orientation constraints)
- Control temperatures of remotely located components (thermal bus)
- Modulate temperatures with changing thermal environment and equipment power
- Low entry mass (most mass located in cruise stage)
- Ease of fine tuning performance after tests
- Versatile thermal control system for future (small, light, cheap) missions

3. PERFORMANCE REQUIREMENTS

After choosing the mechanical pumped cooling loop to serve as the 111{S for M PF, a system level design study was performed on the spacecraft and the following requirements were developed for the HRS:

Thermal:
- Cooling power: 90 -180 w
- Allowable temperature range of equipment: -60 to -20 ^{o}C (low limit), 5 to 70 ^{o}C (high limit)
- Freon liquid operating temperature of + 30 to -10 ^{o}C
- <3 W parasitic heat loss on Martian surface (from any remnants of the cooling loop)

integrated Pump Assembly **(IPA):**
- 0.2 gpm freon flowrate @ > 4 psid pressure rise
- < 10 w total power consumption during cruise
- <8 kg weight
- > 2 years of continuous operation without failure

Leakage:
- Meet specified (very low) leak rate (liquid & gas) to maintain liquid pressure well above saturation pressure (at least 30 psi difference)

Venting:
- Freon to be vented from HRS prior to lander entering Martian atmosphere to prevent contamination of Martian surface (freon would interfere with chemical experiments to be performed by Pathfinder on Mars)
- Freon lines from lander to cruise stage to be cut by pyro cutter after freon has been vented to allow separation of cruise stage from the lander
- Negligible nutation torque of spacecraft due to venting process
- Negligible contaminat ion of spacecraft components during freon venting

4. DESIGN DESCRIPTION AND TRADE-OFFS

The HRS design consisted of six distinct parts:

a) Integrated Pump Assembly (IPA)
b) Freon-11 working fluid
c) HRS tubing
d) l'electronics assembly
e) Freon Vent system
f) Radiator

The primary electronics (the key heat source) is located in the lander basepetal in a highly insulated enclosure. The IPA flows the freon through the HRS tubing from the electronics assembly to the cruise stage radiator. The vent system is used to vent the freon prior to Martian entry.

a) Integrated Pump Assembly:

A schematic of the cooling loop along with the IPA is shown in Figure 2. The IPA has two centrifugal pumps, one of them being the primary whereas the second one serves as the backup in case of failure of the primary; only one pump is on at any time. Each pump (powered by its own motor) produces more than 4 psi pressure differential at 0.2 gpm (0,7 liters/min). The pump/motor assembly has hydrodynamically lubricated journal bearings to minimize bearing wear and frictional power loss, and to maximize the life of the system. Each pump/motor assembly has its individual radiation hardened electronics to power it.

Two wax actuated thermal control valves automatically and continuously split the main freon flow between the radiator and a bypass to the radiator to provide a fixed (mixed) temperature fluid to the inlet of the electronics shelf . this is to account for the continuously decreasing environmental temperature for the radiator on its journey from earth to mars and the constantly changing heat load on the electronics. The thermal control valves use an enclosed wax pellet with bellows to open and close two ports leading up to the radiator and its bypass depending on the freon temperature entering the valve - the set point of the valves is 0 to -7 $^\circ$C which was chosen to be approximately in the middle of the operating temperature limits of the electronics being cooled by the HRS. When freon enters the thermal control valves at temperatures higher than 0 $^\circ$C all the flow is allowed to go through the radiator, whereas when the temperature drops below -7 $^\circ$C all the flow bypasses the radiator - for intermediate values of temperatures, the valve opens partially in each direction.

Four check valves in the IPA prevent the flow from recirculating from the primary (active) pump to the backup (inactive) pump and bypassing of either the electronic.s or the radiator whenever only one pump is on and the thermal control valves are either diverting the flow fully or partially to the radiator. Due the changing environment temperature, the bulk of the freon liquid undergoes a temperature change (-40 to +50 $^\circ$C) during the flight and ground testing - to accommodate this the IPA employs a bellows accumulator to maintain the liquid pressure at least 30 psi above its saturation pressure throughout the flight to prevent cavitation of the centrifugal pumps. The accumulator bellows has a stroke of 24 cubic inches and is sized to account for a liquid volume change of 14 cubic inches due to temperature changes and liquid leaks of as large as 10 cubic inches during the flight (7 months). A detailed design description of the IPA is provided in Ref. 1.

b) Freon-11 Working Fluid:

About fifteen fluids (Ref. 2) were traded-off as candidate working fluids before choosing Freon-11 (CCl_3F) a commonly used refrigerant for building air-conditioners, The working fluid is designed to remain in the liquid phase under all conditions to allow the mechanical pumps to work satisfactorily - this and other considerations lead to several criteria used to trade-off these liquids. Some of the liquids traded-off were: various freons, methanol, ethanol, glycols, Dowtherms and trichloroethylene. The criteria used were:

- Freezing point (less than about -90°C because during the radiator bypass the freon in the radiator could get as cold as -80°C)
- Boiling point (as high as possible to ensure that the operating pressure required to maintain the liquid state is low; also higher than room temperature for ease of handling during ground operations)
- High specific heat and thermal conductivity; low viscosity (for high heat transfer rates and low pressure drops)
- Excellent compatibility with commonly used mat trials like aluminum and stainless st ccl (for long term corrosion proof performance)

The important properties of Freon-1 1 arc:

- Freezing point'- -11 1°C
- Normal boiling point = 24°C
- Vapor pressure at 50°C (highest operating temperature) = 20 psig
- Specific heat = 900 J/kg.]{
- Thermal Conductivity = 0.084 W/m-K
- Viscosity = 0.5 cP
- Density = 1.45 g/cc
- Prandtl Number= 4
- Very compatible with stainless steels; very compatible with aluminum at low moist ure levels (-10 ppm), quite corrosive at high moisture levels (-100 ppm); compatible with some elastomers like VITON and materials like TEFLON

c) Tube Diameters and Materials:

Tube diameters of 1/2", 3/8" and 1/4" were traded-off for heat transfer, pressure drop, pumping power and weight.]/4" was used for the electronics shelf for high heat transfer and the fact that the length there was short (] m) that pressure drop was not excessive. 3/8" tube was used for the radiator because the heat transfer coefficient was not critical in the radiator (large available area, ahout 27 feet long); 3/8" tubing was also used for the transfer lines. Since the radiator and the transfer lines had long lengths of tubing this also minimized the pressure drop in the loop. Freon flow rates were traded-off in terms of heat transfer and pressure drops to come up with an optimum value of 0.2 gpm.

The electronics shelf & radiator use aluminum tubing because the tubing in these zones is brazed to aluminum surfaces which arc used to ensure high heat transfer rates with minimum weight. The transfer lines were made of stainless steel for ease of welding, better compatibility with freon, shorter lengths, and lack of heat transfer requirements.

d) Electronics Shelf Tubing Layout:

Several tubing layouts were investigated to minimize component temperatures, freon pressure drop & pumping power. The key constraints were the temperature limits of the Solid State Power Amplifier (SSPA; 40°C) and the battery (-20 to + 25°C) and the highly localized heating in the SSPA (43 Win a relatively small area). Figure 3 shows the tubing layout used for the electronics shelf in the lander. The cooling loop tubing was strategically routed and wrapped near the high power dissipation area of the SSPA to minimize its temperature rise; the ether electronic boxes (IEM and 1)ST) have a relatively uniform power dissipation and did not require strategic routing of the cooling loop tubing to pick up their heat.

The shelf's facesheet thickness was varied to trade-off heat transfer & mass, Local thickening of facesheet near hot spots was also investigated. A basic thickness of 60 roils for the facesheet (no local thickening) was chosen which satisfies all the thermal requirements. After entry into the Martian atmosphere and landing, the HRS is no longer functional, and the electronics in the lander relics on its thermal mass to manage its temperatures within its limits. Since the SSPA power density is so high, the facesheet was thickened near the SSPA to **180** mils to satisfy the entry and Martian surface requirements (coupling tho high power, low mass SSPA to the low power, high mass IEM box to improve the transient response).

In addition to the lander electronics shelf two other components were cooled by the cooling loop: the Shunt Limit Controller (SLC) and the Rover cold finger. For the Rover the cold finger is coupled to a split, clamshell which grabs onto the HRS tubing to reject, its heat (2 W). The SLC has a heat dissipation varying from O to GO W (depending on the shunted power) and its cooling is achieved by bonding a cold plate Lo its interface - the cold plate has two feet of the cooling loop tubing brazed to it **for** freon flow.

e) Venting:

Before Martian entry, the freon needs to be removed from the lander (to minimize contamination of the Martian surface) by either venting all of it to space or repositioning it to the cruise stage (which is separated from the lander before entry) several schemes to vent the freon were investigated to come Up with a scheme which minimizes the resultant torque on the spacecraft:

Vent freon to space:

- Use high pressure gas (N_2) in the accumulator to "piston-out" freon from the HRS by opening a pyro valve which connect.s the gas side of the accumulator to the liquid - the liquid in turn is vented to space via a nozzle which is opened to space via another pyro valve (Figures 4 and 5)
- Discharge from opposing (T-shaped) nozzles to cancel the torques,
  or, through a single nozzle with the nozzle axis passing through the spacecraft e.g. (with the nozzle outlet pointed in a direction opposite to the e.g.)

Reposition freon from the lander to the cruise stage:

The main reason for the torque on the spacecraft is the reaction from the momentum of the venting freon., hence the rationale for entertaining this possibility because until the spacecraft, is intact (cruise stage connected to the lander), repositioning the freon within the spacecraft should minimize the reactional torque. The scheme was to usc the accumulator gas to push the freon into a separate (extra) thin walled & light weight "holding" tank in the cruise stage (sized to hold the entire volume of liquid freon) - an extra check valve would prevent backflow from the holding tank to the HRS.

Venting freon to space through a single nozzle with its axis passing through spacecraft e.g. was chosen ant] implemented - it is a simple scheme to implement (Figures 4 and 5) with minimum contamination and minimum hardware changes to the spacecraft. The diameter of the nozzle is 1 mm which meets the attitude control system's requirements of the disturbing torque - the time to vent all the freon is predicted to be about three minutes. 'IThc initial thrust from the nozzle is estimated to be about 0.5 N with an initial exit. speed of 21 m/s. 'l'he thrust, of course, decays very rapidly (exponentially) and is less than 0.05 N at the end of the vent process.

f) Radiator:

The radiator used to reject the 180 W of heat (maximum) is a 27' long by 8" wide circumferential strip of aluminum (30 roils thick and thermally at,t.ached to the 3/8" HRS tubing) and located at the circumference of the cruise stage. It is mechanically attached to the cruise stage ribs and thermally (conductively) decoupled by isolators. Both sides arc painted white (NS43G **On** the outside surface, Dexter Crown Metro gloss white on the inside surface; high ϵ, low α) to maximize its heat loss potential. The inside surface is radiatively coupled to the warm cruise stage underside ant] the backshell to preclude freezing of the freon in the radiator when the radiator faces a cold environment. and most of the freon bypasses the radiator @4 '% bypass).

The reason for relying on the radiative coupling instead of conductive coupling to pick up some heat from the cruise stage is that the radiative coupling (and heat. input) is much easier to predict and implement than the conductive coupling because the conductive coupling is via a very convoluted and complex thermal path which also involves contact conductance. For the coldest conditions the cruise stage is at -30°C while the backshell is at -65°C - these surfaces provide enough heat to the radiator in the coldest conditions to maintain the temperature of the coldest portion of the radiator above -80°C, which is well above the freezing point of the freon-1 1 (-1 11°C) - even if there was no freon flow through the radiator, its temperature would not fall below -80°C.

5. PERFORMANCE PREDICTIONS

Radiator:

Table 1 contains the predicts from a detailed multi- -nodal thermal model (SINDA/TRASYS) of the radiator for the worst case hot and cold conditions. The worst hot case occurs near earth at a solar array to sun angle of 30°, whereas the worst cold case occurs near Mars at a sun angle of 41°. The worst combination of thermo-optical properties (highest $\propto$ for all the surfaces) and parasitic heat loads into the HRS tubing (30 W,) were used for the worst hot case. Similarly, the lowest $\propto$ was used to predict the worst cold case properties, even though realistically, near Mars the $\propto$ would have degraded and would be much larger - however, this serves as a conservative approach to bound the predicted temperatures - the nominal cases will exhibit better temperatures. The radiator is sized to be large enough that even in the worst. hot case the radiator is nearly beginning to be bypassed; in the worst. cold case 94% of the flow bypasses the radiator to maintain the correct fluid temperature, In the worst cold case the coldest radiator zone is at -80°C which implies a 30°C margin above the freezing point of freon- 11 (-11 1°C).

Electronics:

A multi-nodal integrate] model of the radiator and electronics (cooled by the pumped loop) was constructed (SINDA/TRASYS) to predict the temperatures of the electronics interfaces for the worst, hot and cold cases (same as for the radiator) - the results are presented in Table 2. All the electronics interfaces meet. their flight allowable limits with margins (all except one component have margins larger than 10°C for the worst hot & cold cases). The maximum heat dissipation for the hot case is 190 W and the minimum is 70 W for the: worst cold case, The temperature rise of the freon through all the electronics is about 1 1°C for 190 W of heat removal at 0.2 gpm.

6. DEVELOPMENT TESTS

Several development tests were conducted to characterize the performance of the cooling loop. These tests were performed in parallel with the design effort and were very helpful to ensure that the final design would meet its requirements, These are described below:

Thermal-hydraulic:

A development test was performed to simulate the electronic shelf and the radiator to validate the thermal and hydraulic performance models used in predicting the performance of the cooling loop. Figure G shows a schematic of the test set-up where the lander electronic shelf's power distribution and tubing layout is simulated. Also simulated were the actual lengths and diameters of the radiator and transfer line tubing. The radiator was not tested thermally (because the heat flux is very small) but tested for pressure drops; the electronics shelf was tested both hydraulically and thermally. The pressure drop across the entire system was measured along with temperatures selected locations on the electronics interfaces. A rotameter was used to measure the flow rates.

Table 3 show's the comparison of predictions and test results - in genera] the prediction were quite consistent with the test results; since the predictions had ample margins this test served its purpose in quantifying the confidence of the thermal-hydraulic design of the cooling loop. The other two components, the rover and the SLC, were not simulated thermally in this test (but were simulated hydraulically) because at the time of performing this test their designs were not complete - they will be tested in the forthcoming spacecraft system thermal vacuum test,

Leaks:

Due to integration constraints 17 mechanical joints (B-Nuts or **AN** fittings) are used to complete the assembly - the rest of the assembly is welded. Any large leaks from the HRS during the **7 month** flight, to Mars would seriously jeopardize the mission. Welded joint were not deemed to leak any significant amount of freon. The B-Nuts, however, being mechanical in nature, could potentially leak and it was considered highly desirable to conduct tests on them to ascertain that they will not leak at rates substantial] enough to deplete the flight accumulator during the mission. It was also desired to come up with better schemes for providing some extra insurance against any potential leaks (epoxying the joints).

An extensive test for assessing the freon leak rate through these mechanical joints (B-Nuts or AN fittings) used in the MPF Heat Rejection System (HRS) was conducted. All the combinations of materials (Al, SS) and sizes **(1/4", 3/8")** used in the flight. HRS were simulated. Teflon flex lines identical to the flight ones were also tested for leaks through their joints. Use of epoxies to provide insurance against leaks was also assessed. Twenty four B-Nut joints were examined. These joints were subjected to cyclic mechanical flexing and torsion to simulate those encountered by the worst joint in the flight system during launch. This was followed by thermal cycling to simulate the excursions during ground testing and flight.

Helium leak tests were conducted on each joint under vacuum and under internal pressure of 100 psia. In addition, all the joints were pressurized with liquid Freon - 11 (used in flight, system) and tested for freon leaks. All the tested joints exhibited leak rates which were much lower than t.hose used to size the flight accumulator - the accumulator is sized to accommodate a leak of 17 cubic inches of liquid freon in the 7 month flight; whereas our tests showed that the total leak should be much lcss than half of this value even under the worst conditions.

Usc of soft cone seals and re-torquing was recommended. Also recommended was the usc of an epoxy on the ext.crier surfaces of the joint s' leak paths to provide additional insurance against, leaks in flight.

Material Compatibility:

Within the HRS Freon-1 1 is in constant contact with materials like aluminum, stainless steel and some elastomers. Concerns for potential corrosion of aluminum, particularly in contact with moist freon, were alleviated by conducting tests to investigate the compatibility of freon-1 1 with **aluminum and** stainless st **ccl, Several** test samples **of** aluminum and stainless **steel were insert** ed **in** freon- 11 with different levels of moisture (freon is supplied in drums at a moisture level of about 10 ppm and it saturates at 100 ppm). These samples were examined chemically, visually and under elect ron microscopes to measure the levels of corrosion as a function of time. For aluminum, no evidence of corrosion was observed for low moisture levels but there was a very strong evidence of corrosion at the high moisture levels. This test showed that it was extremely important to minimize moisture to prevent corrosion of aluminum, and elaborate safeguards were taken in the freon storage & loading process to minimize the moisture levels.

No evidence of corrosion was observed for stainless steel for all the moisture levels tested. VITON **(used** in the check valves) was found to swell significantly when inserted in freon- 11, however, subsequent leak tests performed on the check valve demonstrated that it, the leaks through the check valves in the check direction very small and well within acceptable limits, All other materials in contact with the freon underwent long term compatibility tests and were found to be acceptable.

life:

Since the cooling loop will be used throughout the flight. for seven months (5000 hours) and it crucial to function reliably throughout this duration to guarantee mission success, a life test set up was built and is undergoing long term testing. The schematic of this test, is shown in Figure 7- it simulates the long term operation (5000 hours flight duration) of pump assembly & particle filter, in conjunction with rest of the HRS (Al, Stainless steel, Teflon tubing, accumulator, check valves, etc.). This system has clocked about 4500 hours of uninterrupted operation until now with no pump failures.

The filter used in this mock-up had inadequate capacity and was bypassed after 3600 hours(flight filter has at least 5 times higher capacity for particles). The flight filter uses a check valve to bypass it when the filter's pressure drop is higher than 2.5 psid. Since the I PA produces a pressure riso of more than G psid at 0.2 gpm (the required flow rate for heat transfer), and the pressure drop in the cooling loop system is expected to be 2 psid, this additional pressure drop from a clogged filter should not pose a problem in providing the required flow rate of freon throughout the flight,

] n addition to the compatibility tests described earlier (performed on small sections of tubing materials in a non-flowing environment of freon), this life test was also used to investigate & measure the long term synergistic corrosion of the HRS tubing (Aluminum, Stainless steel) in a flowing environment with all the materials and components used in the flight system simulated. Samples of tubing & freon liquid were taken out periodically for analysis - no evidence of corrosion has been found until now.

This life test was also used to measure long term leaks from the HRS, particularly due to mechanical joints ("AN" fittings, "II-Nut s") - relatively large leaks were observed in the beginning of test which were corrected ant] prompted a more elaborate leak test done separately (discussed earlier).

Figure 8 shows the variation in the flow rate, pressure drop and pump input power as a function of time for this life test. Note the sharp change in all these parameters after the filter was bypassed.

7. CURRENT STATUS AND NEAR TERM PLANS

The fully assembled MPF spacecraft has successfully undergone acoustic, and radiated emission and susceptibility tests; the 1 IRS has been integrated with the spacecraft, and will under-go extensive testing in the forthcoming System Thermal Vacuum test. The radiator (12 panels on cruise stage), electronics cooling assemblies, integrated pump assembly have been integrated with rest of the spacecraft. The life test has undergone 4500 hours of operation (500 more for completing the flight duration) - it will be run for at least one more year (beyond spacecraft launch) and will servo as a test bed for future cooling loops. The launch is scheduled for Dec '9G followed hy landing on Mars in July '97.

8. CONCLUSIONS AND NEXT GENERATION LOOPS

A successful performance of this system in the Pathfinder mission will remove some reluctance (and the associated paranoia) on the part of projects to try out bold new approaches to thermal control of spacecraft. The current Mars Pathfinder HRS life test fluid cooling loop serves as a test bed for developing better cooling loops for future spacecraft. Even though the current loop is designed for Pathfinder, the technology developed will be of generic use for future missions. The knowledge gained will be useful to develop more flexible, robust, compact, lightweight, modular and autonomous mechanically pumped cooling loops. examples would be the development of accumulators which can accommodate large system leaks, different operating fluids, better temperature modulation schemes, gathering of more life cycle data (e.g. pumps with or without filters) to develop more optimal strategies for component redundancies,

9. ACKNOWLEDGEMENTS

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10. REFERENCES

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MARS PATHFINDER HEAT REJECTION SYSTEM

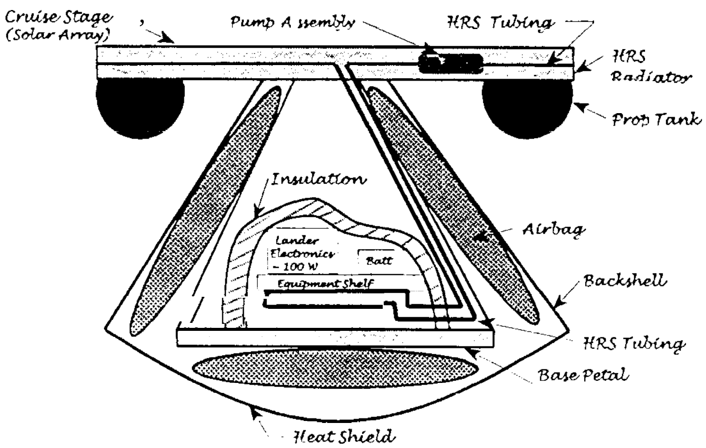

Figure 1

Integrated Pump Assembly

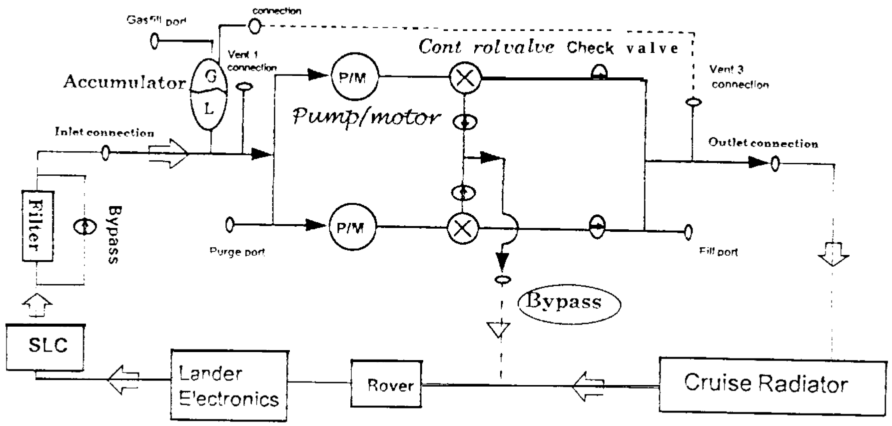

Figure 2

ELECTRONICS SHELF TUBE LAYOUT

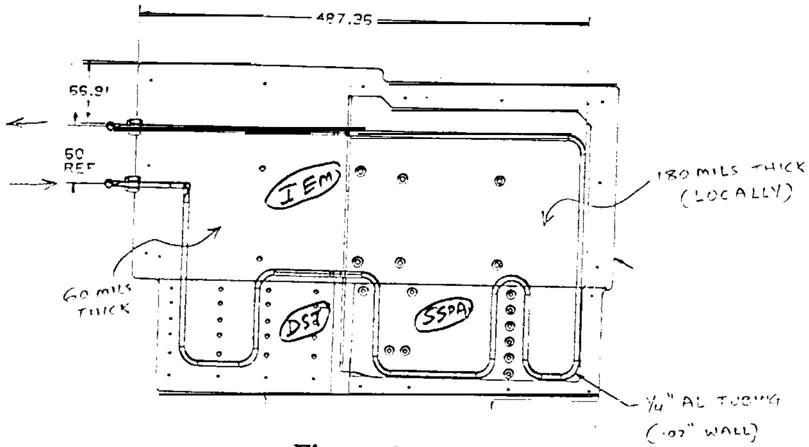

Figure 3

HRS DEVELOPMENT TEST

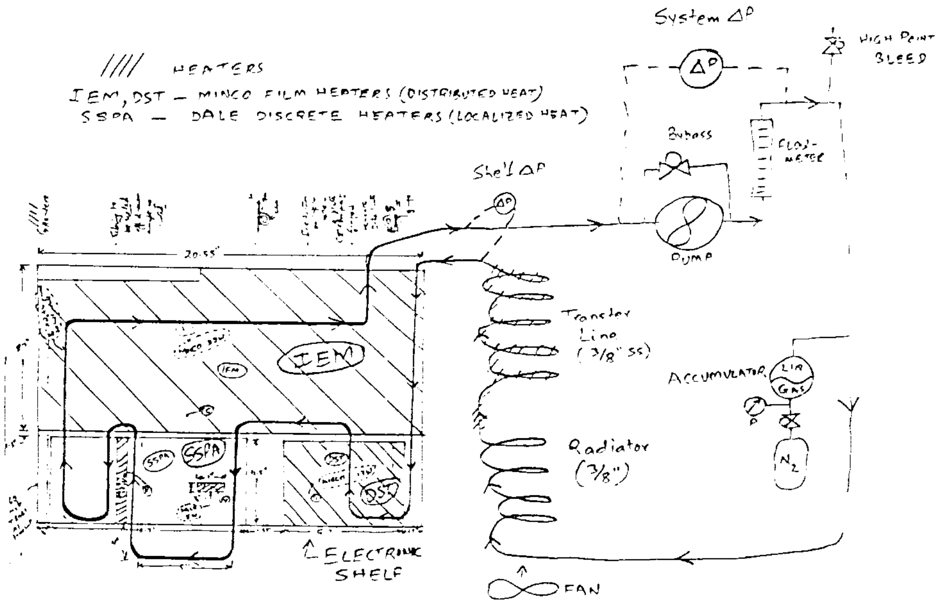

Figure 6

THERMAL-HYDRAULIC DEVELOPMENT TEST

ZONE	PREDICTION (°C)	TEST (°C)	MAX. FLIGHT ALLOWABLE (°C)
SSPA (ISLAND)	14	15	40
SSPA (STRIP)	19	11	40
IEM	15	9	50
DST	5	7	70

ZONE	PREDICTED (psid)	MEASURED (psid)	PUMP RISE (psid)	MARGIN
SHELF ONLY	0.45	0.5		
ENTIRE LOOP	0.9	1	4	300 %

(@ 0.2 gpm)

Table 3

Life Test Schematic

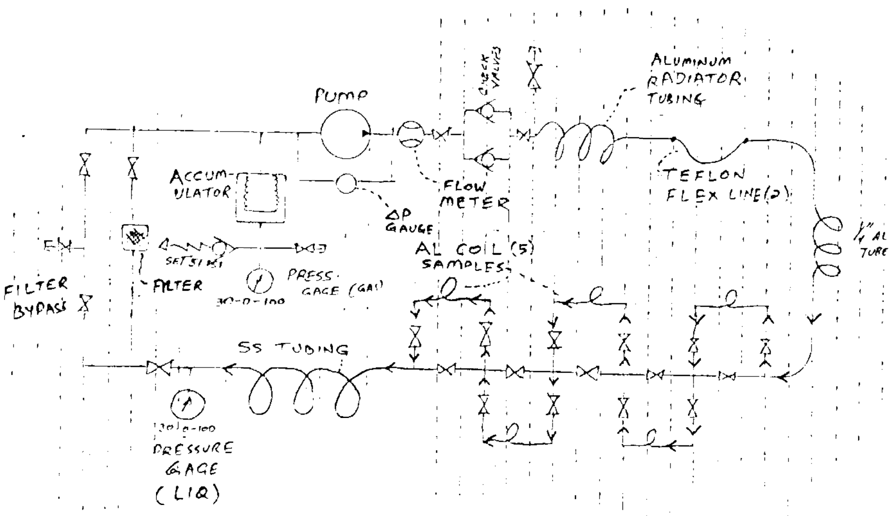

Figure 7

HRS LIFE TEST PERFORMANCE

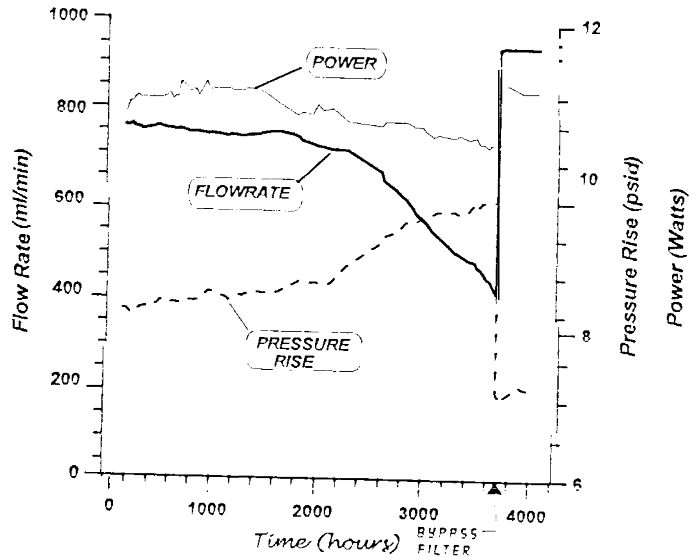

Figure 8